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Railway Applications — Wheel-rail contact geometry parameters — Technical report and background information about EN 15302

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TECHNISCHER BERICHT

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Bahnanwendungen - Rad-Schiene-Berührgeometrieparameter - Technischer Bericht und Hintergrundinformationen zur EN 15302

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Contents

Page

European foreword 4
1 Scope 5
2 Normative references 5
3 Terms and definitions 5
4 Overview of the most important changes made to EN 15302 6
4.1 List of main changes 6
4.2 Additional wheel-rail contact geometry parameters 6
4.2.1 Rolling radii coefficient 6
4.2.2 Nonlinearity parameter 6
4.3 Methods for evaluation of equivalent conicity 7
4.4 Assessment of the smoothing process 7
4.5 New assessment of the complete process 8
5 Technical background to and justification of changes in the revised EN 15302 8
5.1 Equivalent conicity 8
5.1.1 Review of equivalent conicity results obtained with different software tools 8
5.1.2 Comparison with multibody system simulation results 11
5.1.3 Influence of discretisation step size of the rolling radius difference function 14
5.2 Rolling radii coefficient 15
5.2.1 Background 15
5.2.2 Current method 17
5.3 Nonlinearity parameter 20
5.4 Calculation of equivalent conicity by two-step integration 22
5.5 Calculation of equivalent conicity by direct integration of the kinematic equation of motion 23
5.6 Calculation of equivalent conicity by harmonic linearization 23
5.7 Updated reference profiles and results based on analytical solutions 25
5.8 Revised assessment of the smoothing process 27
5.9 Example for uncertainty assessment of the complete process 27
5.10 Influence of simplifications 31
5.10.1 General 31
5.10.2 Wheelset roll movement (rotation around the longitudinal axis) 31
5.10.3 Contact elasticity of wheel and rail 36
6 Guidance for the application of the wheel-rail contact parameters given in EN 15302 39
6.1 Fields of application – Overview 39
6.2 General guidelines 39
6.3 Selection of appropriate reference profiles for assessment of rail head profiles and/or wheel profiles 40
6.3.1 General 40
6.3.2 British Rail Research Survey 40
6.3.3 Reference profiles in the DynoTRAIN project 40
6.3.4 Assessment of design wheel profiles and design rail profiles 42
6.4 Development of equivalent conicity of wheelsets over mileage 43
6.5 Assessment of the contact geometry of a line 45
6.5.1 Methods for determining averaged contact geometry parameters 45

Documents
Document Sharing Hub
Full document details: PD CEN/TR 17792:2022

6.5.2 Assessment of a line for different wheel profiles 46
6.6 Rolling radii coefficient and radial steering index 48
6.7 Nonlinearity parameter 51
6.8 Equivalent conicity in wheel-rail maintenance and interface with TSIs 53
6.8.1 General 53
6.8.2 Equivalent conicity that a vehicle was designed and tested for 53
6.8.3 Equivalent conicity as parameter in wheel profile maintenance regimes 53
6.9 Clarification of wheel-rail contact test conditions according to EN 14363 54
6.10 Application of Contact angle parameter and Roll angle parameter 55

7 Alternative contact parameters not handled in the standard 55
7.1 Difference of contact angles and gravitational stiffness 55
7.2 Contact Concentration Index 56

8 Approximation of equivalent conicity by simple alternative methods 60
8.1 Background 60
8.2 British Rail Research investigations 60
8.2.1 Initial BRR work in 1980s 60
8.2.2 BRR further work in 1990s 63
8.3 Investigations on Quick conicity using DynoTRAIN data 66
8.3.1 DynoTRAIN project data collection 66
8.3.2 Investigations on rail data 67
8.3.3 Investigations on wheel data 73
8.3.4 Combined assessment – track and wheelset 75
8.3.5 Next Steps 75
8.4 Ongoing development of Gradient Index Profile (GIP) 76
8.4.1 Definition of GIP 76
8.4.2 Comparison between equivalent conicity and GIP combined 77

9 Development and usage of the so called conicity maps 77

10 Plausibility check of measured profiles and elimination of outliers 79
10.1 Introduction 79
10.2 Profile area to be covered 79
10.3 Spacing of points on the profile 79
10.4 Elimination of outliers 80

11 Examples for validation of profile measuring systems 81
11.1 General 81
11.2 Evaluations of rail profile measuring systems 81
11.3 Evaluations of ground-based wheel profile measuring systems 83

12 Effect of wheel diameter differences on the running behaviour 84

Bibliography 85

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Document Sharing Hub

Full document details: PD CEN/TR 17792:2022

European foreword

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1 Scope

This document provides background information regarding the changes from EN 15302:2008+A1:2010 to the revised version dated 2021, including the reasons for decisions and additional explanation and guidance that is not appropriate in the standard.

The range of equivalent conicity results obtained with different software tools is described. The additional wheel-rail contact parameters, rolling radii coefficient and nonlinearity parameter, are explained. More information is also provided on the different calculation methods and the updated reference profiles for the assessment. The influence of simplifications used in determination of equivalent conicity is discussed.

To provide more information on the importance of considering the complete measurement and calculation process, methods for plausibility checks, eliminating outliers and assessing the uncertainty and repeatability of measurements are included as well as assessments of the smoothing process.

Guidance is given on fields of application of the wheel-rail contact parameters, on the selection of appropriate reference profiles (choice of reference rail profile and rail inclination for assessing wheel profiles and vice versa) and on handling special cases.

As some references in EN 14363 to wheel-rail contact test conditions have caused difficulties in understanding, clarifications issued by ERA are mentioned.

Interpretation of equivalent conicity results, using tools such as conicity maps, is discussed and various approximations such as ‘quick conicity’ assessments are also described.

Information is included on possible additional wheel-rail contact parameters, not yet ready for standardization, but where further experience is needed.

NOTE In this document the commonly used term “wheel-rail contact geometry” is used as a synonym for the more precise term “wheelset-track contact geometry”.

2 Normative references

There are no normative references in this document.

3 Terms and definitions

No terms and definitions are listed in this document.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

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4 Overview of the most important changes made to EN 15302

4.1 List of main changes

The list below provides an overview of the main changes introduced in the revised EN 15302:

— extension of the Scope;

— introduction of new wheel-rail contact geometry parameters (rolling radii coefficient, nonlinearity parameter);

— improvement of the description of the methods for evaluation of equivalent conicity including the determination of the lateral peak displacements;

— introduction of additional methods for evaluation of equivalent conicity;

— improvement of the description of the reference profiles;

— introduction of the additional reference wheel profile C;

— reference results based on analytical solutions;

— hints for plausibility checking of measured wheel and rail profiles;

— revised assessment of the profile smoothing process;

— new assessment of the complete process for determination of wheel-rail contact parameters.

In this Technical Report the ideas behind the mentioned changes and a more detailed explanation are given where necessary.

4.2 Additional wheel-rail contact geometry parameters

4.2.1 Rolling radii coefficient

In addition to the now well-established parameter “equivalent conicity”, which describes the contact geometry in straight track and in curves with very large radii based on a simplified model of the run of the wheelset, an additional parameter for the guiding behaviour of the wheelset in curves with small and very small radii is defined. This parameter, the so-called rolling radii coefficient, is intended to describe the capability of achieving a radial position of a wheelset in the curve. Details are given in 5.2 and 6.6.

4.2.2 Nonlinearity parameter

Equivalent conicity is traditionally used to assess the wheel-rail contact geometry in regard to running stability. However, the equivalent conicity as a linearized parameter does not consider the nonlinearity of wheel-rail contact geometry. One value of equivalent conicity is usually used to characterize the wheel-rail contact geometry: the equivalent conicity value for a wheelset displacement amplitude of 3 mm. However, the same value of equivalent conicity for a wheelset displacement amplitude of 3 mm can arise from a large number of very different contact geometries, see Figure 1.

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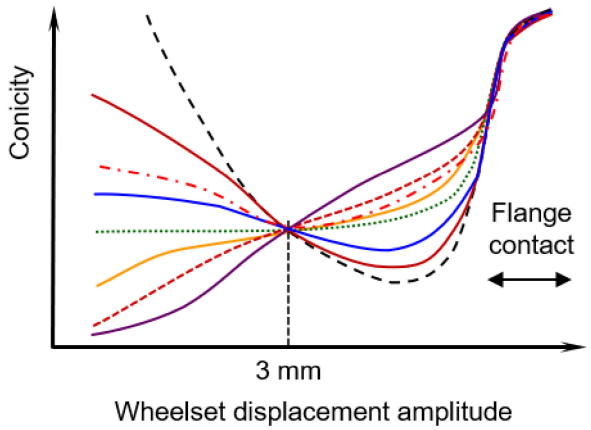


Figure 1 — Possible equivalent conicity functions determined from a set of wheel-rail contact geometries with the same equivalent conicity value for a wheelset displacement amplitude of 3 mm.

Simulation studies [1] and [2] demonstrated, that the vehicle's dynamic behaviour at the stability limit depends on the overall properties of the wheel-rail contact geometry; therefore, also on the overall shape of the equivalent conicity function for a range of wheelset displacements inside of the clearance between wheelset and track (i.e. before flange contact).

A second parameter called nonlinearity parameter is proposed in [2] to enhance the characterization of the wheel-rail contact geometry. This parameter represents the slope of the conicity function between the wheelset amplitudes of 2 mm and 4 mm. The nonlinearity parameter does not replace the equivalent conicity as used for the characterization of wheel-rail contact geometry regarding the stability. It should be understood as additional information complementing the equivalent conicity. While the equivalent conicity value for a wheelset amplitude of 3 mm represents a "level parameter" for the assessment of contact geometry regarding the instability limit according to EN 14363, the nonlinearity parameter has to be understood as a "performance parameter", characterizing the vehicle performance at the stability limit as well as the sensitivity of vehicles to the lateral excitation by track irregularity. Details are given in 5.3 and 6.7.

4.3 Methods for evaluation of equivalent conicity

The description of all evaluation methods was largely improved. All calculation steps are now explained. In particular, the two-step integration method was clarified (see 5.4 for details), and a description of the direct integration of the differential equation has been added (see 5.5 for details).

Moreover, it is pointed out that the linear regression and the harmonic linearization (see 5.6 for details) are approximations, which may give good results but have to be used with care.

Harmonic linearization has been developed in the 1970s to determine linearization parameters required for linearized calculations of railway vehicle dynamics. As the method is usually available in simulation tools, it is also used for the determination of equivalent conicity of measured profiles of wheels and rails. It was thus decided to include this method in the current revision of the standard EN 15302.

4.4 Assessment of the smoothing process

As in the former versions of EN 15302, the effects of profile errors originating from the profile measurement still have to be assessed. However, the definition of the errors to be used for the assessment is revised and updated according to the performance of current measuring systems as well as of the increased available computation power. Further, new quality numbers for the equivalent conicity and the rolling radii coefficient are introduced describing the ability of the tested smoothing algorithms to deal

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with measuring errors. Hence it can be checked if the smoothing process meets the requirements taking the measuring accuracy of the used profile measuring system into account.

More details are provided in 5.8.

4.5 New assessment of the complete process

According to EN ISO 10012:2003 (Measurement management systems - Requirement for measurement processes and measuring equipment), an effective measurement management system ensures that measuring equipment and measurement processes are fit for their intended use and is important in achieving product quality objectives and managing the risk of incorrect measurement results.

An important part in/of the measurement management system is the metrological confirmation including estimation of measurement uncertainty. The commonly used method for the estimation of measurement uncertainty is described in ISO/IEC Guide 98-3:2008 - Guide to the expression of uncertainty in measurement (GUM: 1995). A measurement cannot be properly interpreted without knowledge of its uncertainty.

Corresponding to these standards a new assessment method for the complete process of wheel-rail contact parameter determination (including measurement and calculation) is introduced in EN 15302. In 5.9 of this Technical Report an example is given for the possibility of estimation of measurement uncertainty applied to the wheel-rail contact parameters derived from measured rail profiles.

The different methods applied today for assessment of measuring uncertainty are at least as strict as the requirements used when the current limit values for wheel-rail contact parameters were established. The limit values already include a margin for measuring uncertainty and no additional adjustment of the result or the limit value shall be made.

5 Technical background to and justification of changes in the revised EN 15302

5.1 Equivalent conicity

5.1.1 Review of equivalent conicity results obtained with different software tools

In the beginning of the revision of EN 15302 a benchmark comparison of currently used calculation methods for equivalent conicity tan γ_e was carried out in order to check the tolerances given in the Standard against the methods. The test included all combinations of the reference wheel profiles with the reference rail profile A as defined in the EN 15302:2008+A1:2010 as well as a selected wheel-rail combination representing the special case described in B.3 of that document (hollow worn wheel profile). The tan γ_e functions have been calculated for the following methods:

— direct integration of the differential equation of lateral wheelset motion;

— harmonic linearization;

— two-step integration as described in EN 15302:2008+A1:2010, Annex B;

— linear regression as described in EN 15302:2008+A1:2010, Annex C;

— analytical solution (where applicable).

In some cases, the methods are applied also accounting for the elasticity in the wheel-rail contact (non-elliptical contact patches) and/or the effect of the axle's roll angle around the axis longitudinal to the track due to the lateral shift of the wheelset. All the tested methods are implemented in at least two different software tools. In total the calculation results listed in Table 1 have been provided for the benchmark.

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Table 1 — Available results for equivalent conicity

Identifier	Method	Roll angle considered	Elastic contact
DB Netz	Direct Integration	No	No
ITCF (DMA)	Direct Integration	No	No
ALSTOM	Two-step Integration	No	No
SNCF (Klingel)	Direct Integration	No	No
SNCF (Ann. C)	Linear Regression	No	No
SNCF (SIMPACK)	Direct Integration	?	No
Siemens (integ.)	Direct Integration	No	No
Siemens (Ann. B)	Two-step Integration	No	No
Siemens (Ann. C)	Linear Regression	No	No
Siemens (harmonic)	Harmonic Linearization	No	No
Siemens (RSGEO)	Harmonic Linearization	Yes	No
Siemens (SIMPACK integ.)	Direct Integration	No	Yes
Siemens (SIMPACK harm.)	Harmonic Linearization	No	Yes
DB Systemtechnik	Two-step Integration	No	No
IIR (ETQ)	Linear Regression	Yes	No
IIR (Vampire)	Linear Regression	Yes	No
NR	Two-step Integration	No	No

The calculation results of the different methods are shown in the following Figures together with the reference results and the respective tolerances according to EN 15302:2008+A1:2010, Annex F. Figure 2 contains the results for the symmetrical cases (identical profiles and identical wheel diameters at left- and right-hand side) whereas Figure 3 provides the graphs for the cases with a wheel diameter difference of 2 mm and Figure 4 for the asymmetrical wheel profiles. The analytical solutions are not plotted here because they are nearly identical to the related original reference results.

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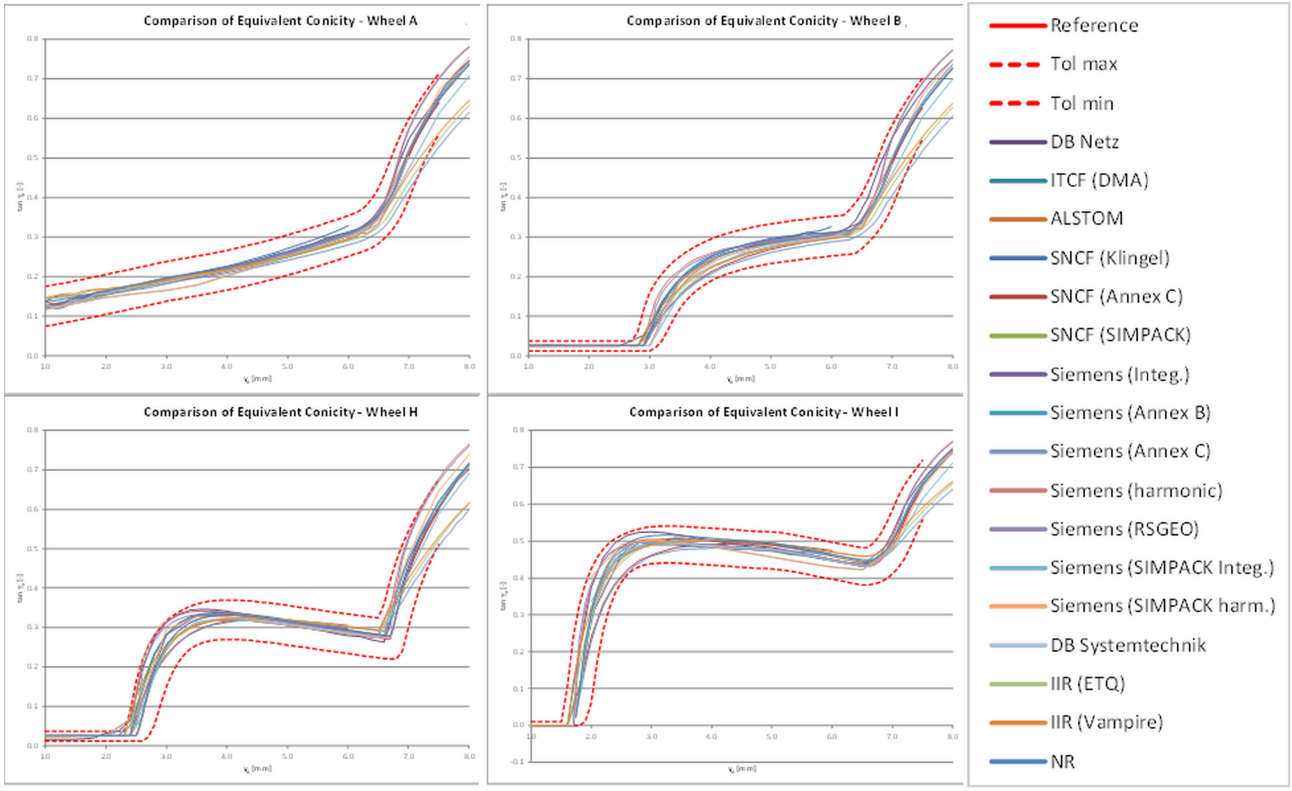


Figure 2 — Calculation results for equivalent conicity of various calculation methods (reference profiles in nominal condition)

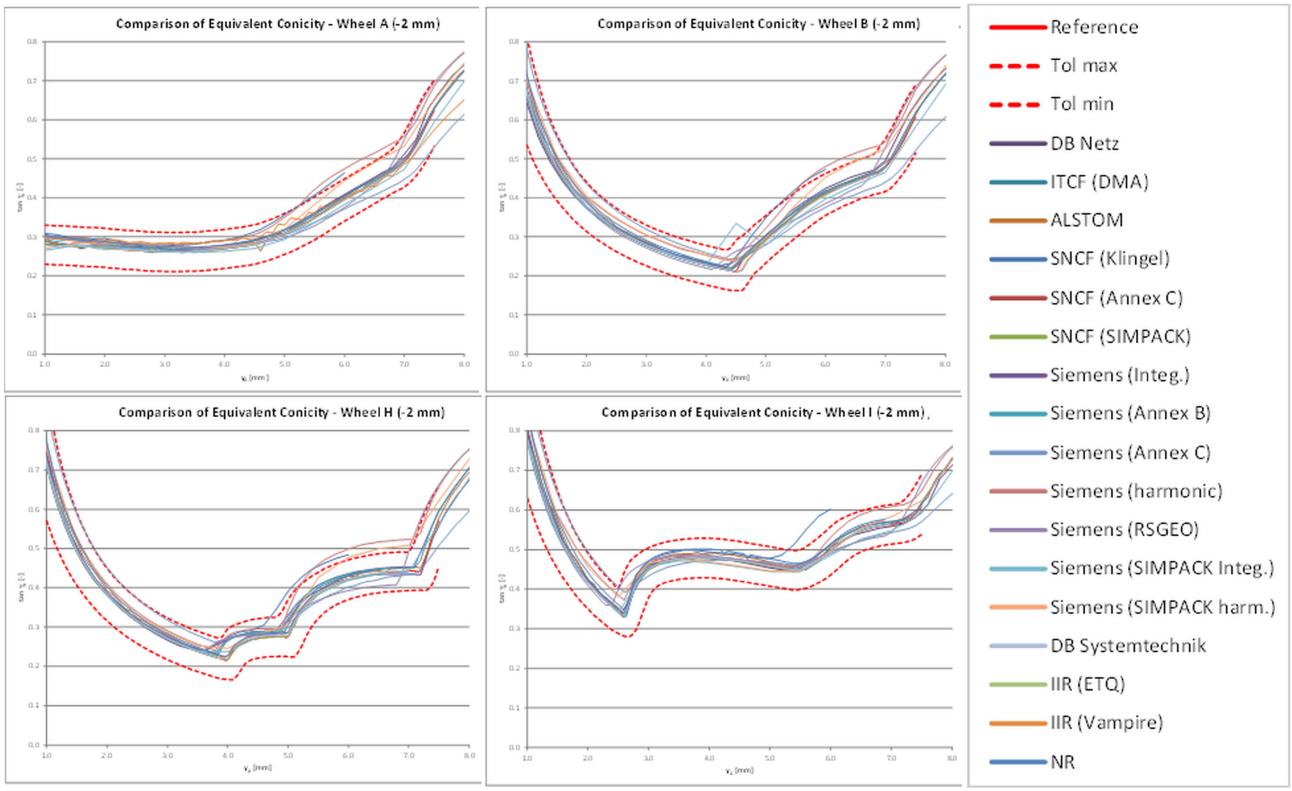
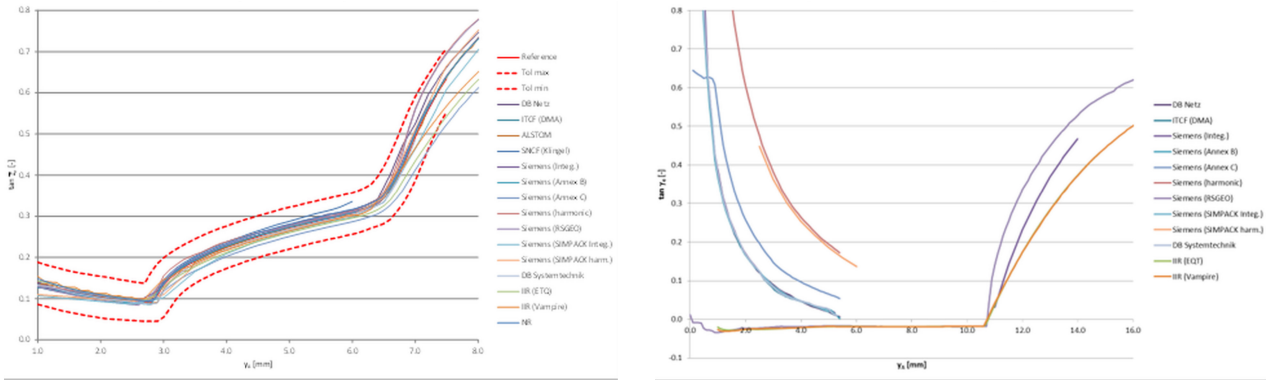


Figure 3 — Calculation results for equivalent conicity of various calculation methods (wheel diameter difference of 2 mm applied)

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a) Comparison of equivalent conicity wheels A+B

b) Comparison of equivalent conicity worn wheel

Figure 4 — Calculation results for equivalent conicity of various calculation methods (asymmetrical wheel profiles)

Except for the wheel-rail combination representing the special case described in B.3 (right diagram in Figure 4), the comparisons show good agreement of the different methods and also confirm that the tolerance bands for the equivalent conicity as given in EN 15302 are practical. There are only a few methods providing results partly outside the tolerances, mainly for large lateral wheelset amplitudes where the contact position is at or close to the wheel flange. As the practical meaning of equivalent conicity values for this range of lateral wheelset amplitudes is very limited (see also below) it was decided to restrict the normative range for which a new calculation method shall be tested against the reference results to amplitudes of 1 mm to 6 mm.

The performed investigation showed also the high importance of a unique definition of the lateral wheelset displacement. In the beginning, for some methods the lateral wheelset displacement was measured at the centre of gravity of the wheelset. In combination with the consideration of the roll movement around the longitudinal axis this resulted in significant deviations of the equivalent conicity functions. Therefore, the revised EN 15302 contains a clear statement now: “the lateral displacement of the wheelset as used in this document is considered at the top of rail level”.

The large scatter of conicity results for the special case with the hollow worn wheel, see the right diagram of Figure 4, showed that there is a need for more information on how to deal with such cases. Therefore, a new Annex H has been added to EN 15302 explaining the possible existence of multiple solutions. It is also important to understand that the negative values of equivalent conicity shown by some calculation tools have no physical meaning.

5.1.2 Comparison with multibody system simulation results

In order to find out up to which lateral displacement the obtained kinematic wheelset movement provides a physically reasonable assessment, multibody system (MBS) simulations have been performed and the resulting wavelengths of the lateral wheelset motion have been compared with the wavelengths of the respective kinematic wheelset trajectory. The dynamic solutions for the lateral wheelset motion are found by means of simulations of a single vertically loaded wheelset with a soft primary suspension moving along straight track. Starting with an initial lateral displacement the lateral wheelset trajectory is calculated and analysed. The equivalent conicity is calculated based on the changing wavelength according to Klingel's formula and plotted against the related wheelset amplitude, see Figure 5.

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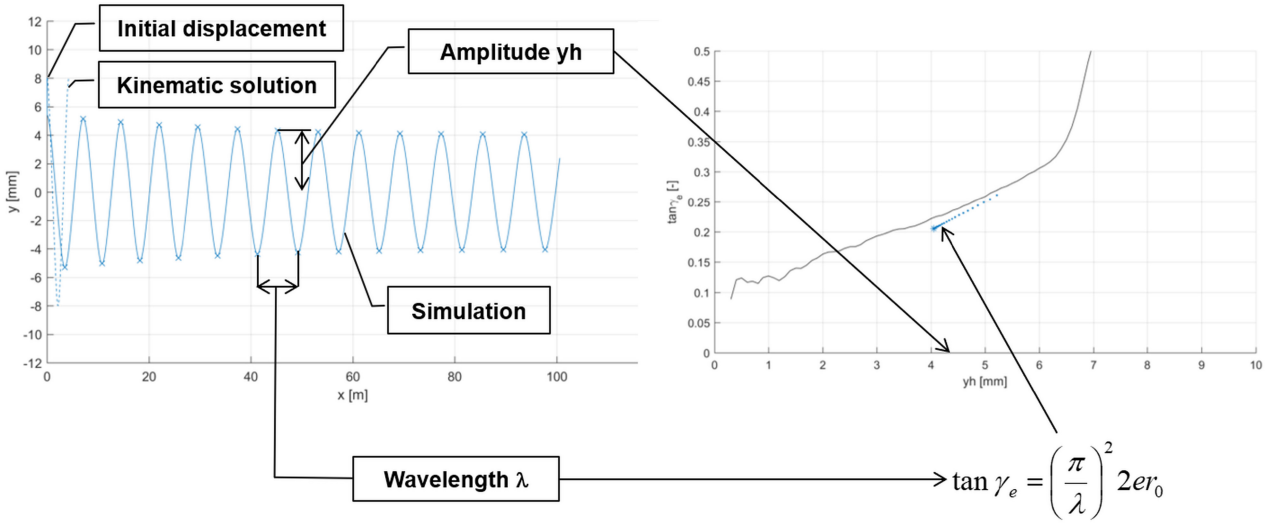
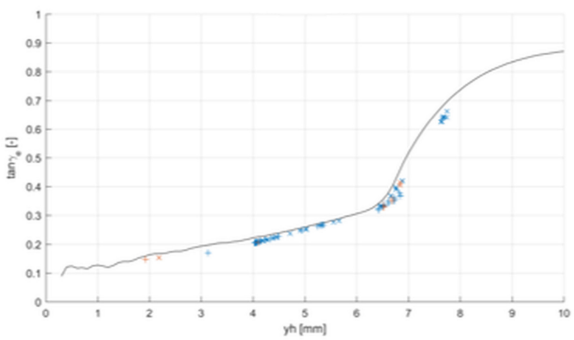
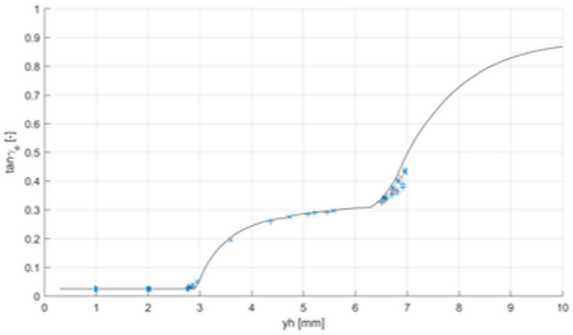


Figure 5 — Determination of equivalent conicity by means of MBS simulation

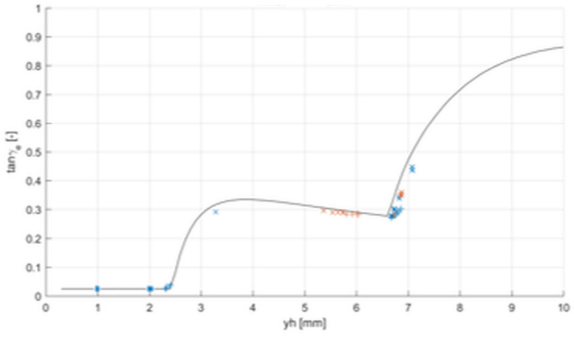
By varying the input parameters speed, primary stiffness (in the range below 2e6 N/m) and wheel-rail friction coefficient a wide range of amplitudes has been covered. In the following Figures the resulting conicity values (coloured markers) are compared to an example of the kinematic solution (solid line) for all the reference cases of EN 15302.



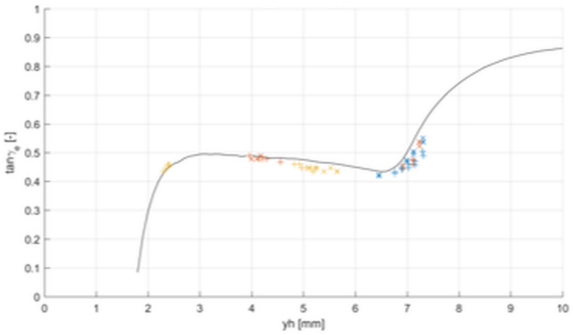
a) wheel A/rail A



b) wheel B/rail A



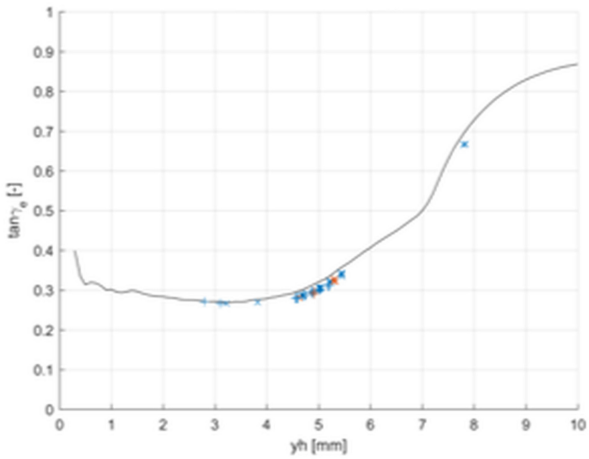
c) wheel H/rail A



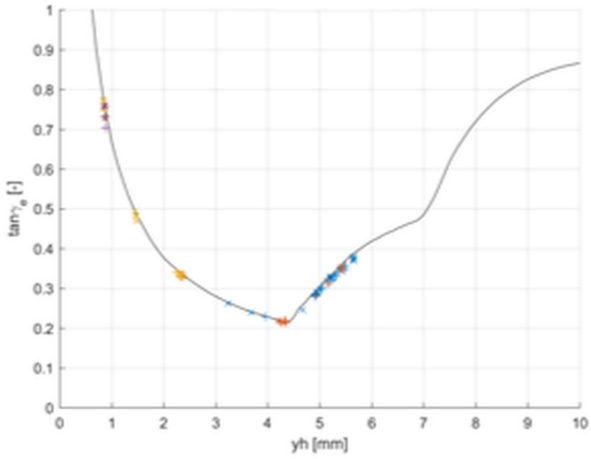
d) wheel I/rail A

Figure 6 — Comparison of kinetic and kinematic solutions for equivalent conicity (reference profiles in nominal condition)

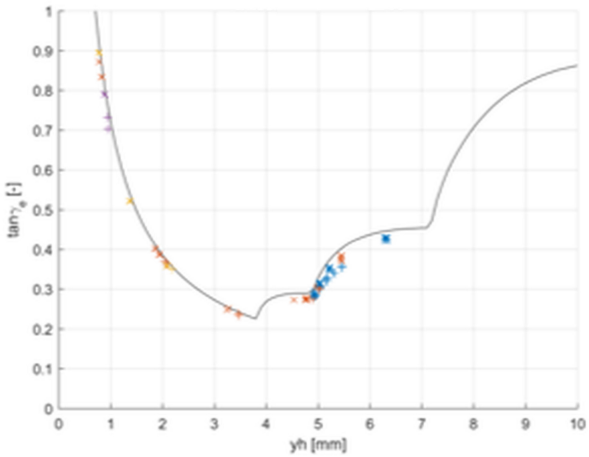
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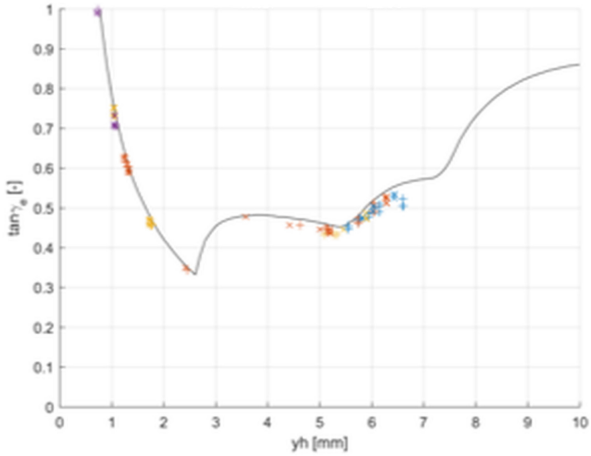
a) wheel A (ΔD = 2 mm)/rail A



b) wheel B (ΔD = 2 mm)/rail A



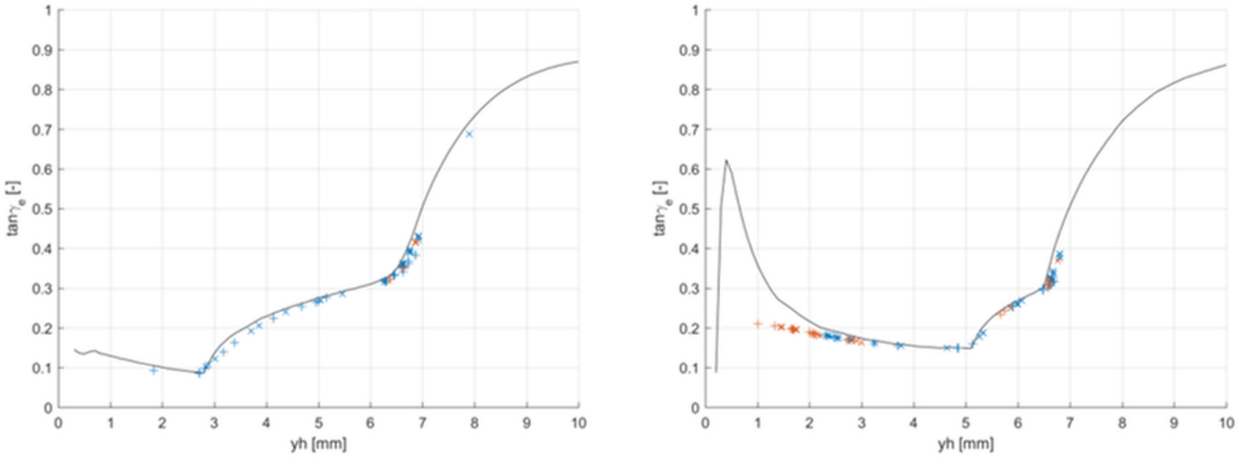
c) wheel H (ΔD = 2 mm)/rail A



d) wheel I (ΔD = 2 mm)/rail A

Figure 7 — Comparison of kinetic and kinematic solutions for equivalent conicity (wheel diameter difference of 2 mm applied)

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a) wheels AB/rail A **b) S1002–1425/60 E1, 1:40, 1435**

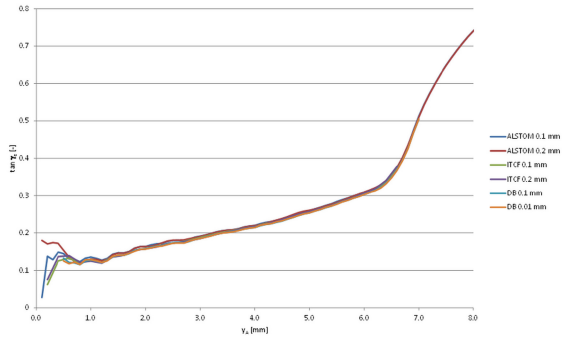
Figure 8 — Comparison of kinetic and kinematic solutions for equivalent conicity (asymmetrical case of EN 15302 and combination of wheel profile S1002 with rail profile 60E1)

It can be concluded for the selected parameters that the MBS simulation of the wheelset movement gives very similar results to the kinematic solutions. In most cases the wavelength is a bit longer leading to slightly lower equivalent conicity values. However, in some cases of large amplitudes (covering large contact angles) or big jumps of the contact point positions no dynamic solution for the harmonic wheelset oscillation was found. This confirms the chosen range of amplitudes (1 mm to 6 mm) for testing calculation tools for equivalent conicity.

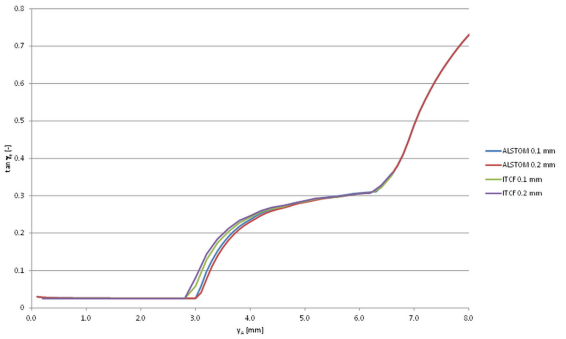
5.1.3 Influence of discretisation step size of the rolling radius difference function

Another question raised during the work on the revision of EN 15302 is related to the influence of different discretisation step sizes of the rolling radius difference function Δr(y) on the equivalent conicity. In addition to the results shown above some workgroup members provided conicity results for different sampling rates of the Δr-function varying between 0,01 mm and 0,2 mm. Based on these results (presented in Figure 9) it can be concluded that the step size used for Δr(y) may be important if only a single conicity value at a wheelset amplitude y_A close to a jump in the conicity function is considered. Nevertheless, for practical applications it is sufficient to use a step size of 0,2 mm as long as only the equivalent conicity is of interest. However, for the determination of the rolling radii coefficient a maximum step size of 0,1 mm is required.

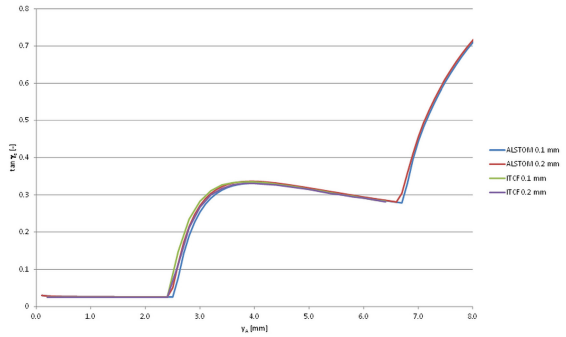
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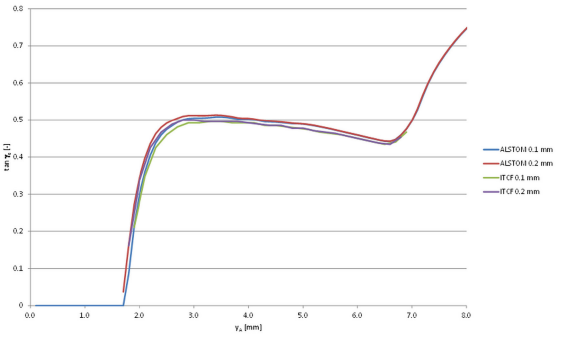
a) Comparison of equivalent conicity wheel A/rail A



b) Comparison of equivalent conicity wheel B/rail A



c) Comparison of equivalent conicity wheel H/rail A



d) Comparison of equivalent conicity wheel I/rail A

Figure 9 — Influence of the discretisation of the Δr-function on equivalent conicity

5.2 Rolling radii coefficient

5.2.1 Background

The many years of experience of many railways have confirmed the importance of recording and assessing wheel-rail contact geometry, especially for high-speed traffic. The procedure for determining the parameters and functions of contact geometry in straight lines and in curves with very large radii has been regulated by the UIC 519 [21] and incorporated in European Standard EN 15302. The UIC 518 [22], which deals with the running test specifications for vehicle homologations, specifies the requirements for the contact geometry parameters. At this time these requirements related exclusively to straight lines and curves with very large radii. Therefore, the goal was to define a relevant parameter based on a simplified model for the guidance behaviour of the wheelset in curves with small and very small radii. This parameter shall describe the capability of achieving a radial position for a wheelset in the curve.

The influence of the contact geometry on the dynamic behaviour of a vehicle in curves with small and very small radii has been studied several times in the literature. Running through curves was also a subject of both the theoretical and experimental investigations of ORE B 176. One of the essential conclusions of these investigations is the recognition and confirmation that not only the adhesion coefficients but also the contact geometry significantly influences the running behaviour of a vehicle in curves. The theoretical considerations were presented in ORE B 176/DT 292, Part 1 [19]. The results of the experimental investigations were documented in ORE B 176/DT 292, Part 5 [20].

The statement in [19] "... the continuous measurement of the rail profile geometry and thus of the wheel-rail contact is currently not possible ..." corresponded to the then state of the art in the field of recording rail profiles. Modern measurement technology now makes it possible to record the rail profiles from

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measuring vehicles at very short intervals and at reasonable speeds of the measuring vehicle. Thus, extensive new measurement results are available that have been further processed and assessed.

The evaluation of the contact geometry in straight lines and in curves with very large radii according to UIC 519 [21] provides the parameters and functions:

— equivalent conicity;

— Δr function;

— Σtanγ function;

— position of the contact points on the wheel and on the rail.

These parameters and functions also form the basis of the analysis of the contact geometry in curves with small and very small radii. The equivalent conicity has only a relevance for such vehicles which, due to the design of the running gear, allow for sinusoidal running in curves with smaller radii.

The aim of an optimal, track friendly vehicle design should be to pass through a curve with the lowest possible wheel-rail forces and thus to minimize wear and rail fatigue. The profile shape of the wheels which is based on a conical profile, so that the wheelset corresponds to a double cone, allows for pure kinematic rolling in the curve. This wheelset rolling in the curve is characterized by the so-called radial position of the wheelset which is corresponding to an angle of attack = 0. There are three types of wheelset position in the track:

— over-radial steering of the wheelset;

— under-radial steering of the wheelset;

— radial steering of the wheelset (angle of attack = 0).

The different wheelset positions are influenced not only by the contact geometry parameters but also by the vehicle suspension design and the wheel-rail friction conditions. For example, lubrication of the flange contact on the outer rail may reduce the effectiveness of radial steering.

ORE B 176/DT 292, Part 5 [20] defines parameters describing the capability of achieving a radial position of the wheelset which are obtained from the points O, A^{o}, A^{oo} of the Δr function, see Figure 10.

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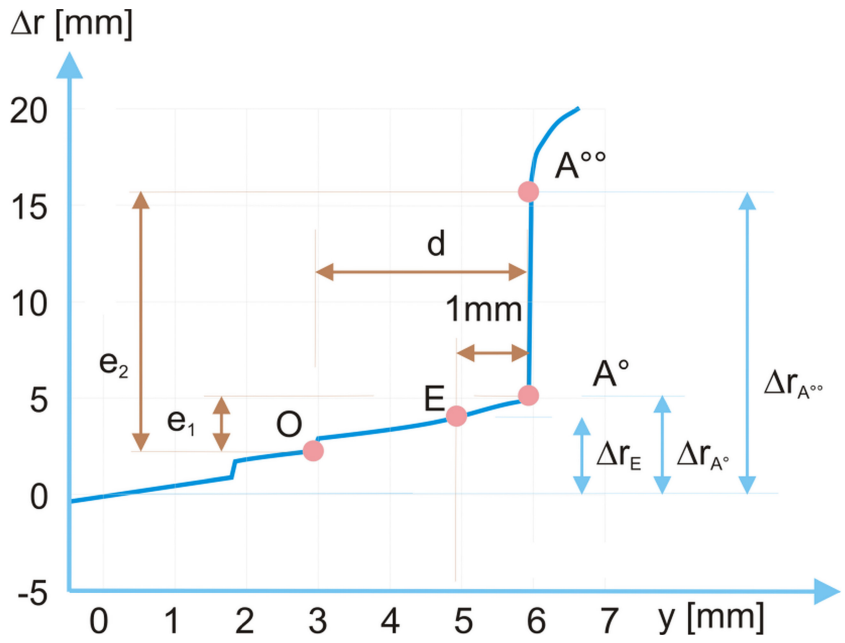


Figure 10 — Definition of parameters for running in curves based on the Δr function

5.2.2 Current method

A free wheelset runs optimally through a curve if it can pass purely rolling without any sliding movement. The two relevant requirements for this are:

1) Movement of the wheelset on the so-called kinematic rolling line

The trajectory corresponds to the lateral offset of the wheelset at which the rolling radius difference Δr amounts to

$$\Delta r = r_0 \frac{2b_A}{R}$$

with:

— $2b_A$ ... nominal contact point spacing (1 500 mm for standard gauge);

— r_0 ... radius of the wheel when the wheelset is centred on the track;

— R ... curve radius.

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2) One-point contact on the outer wheel

The functions Δr = f(y) and Σtanγ = f(y) of a wheel-rail profile combination are very important for the contact geometry in curves. An ideal contact geometry from the point of view of the general curving behaviour is present for a given wheel-rail-profile combination if the Δr and Σtanγ functions provide:

— a smooth continuous function;

— no offset at the central wheelset position in the track;

— no asymmetry with respect to the direction of the curve.

If there is a discontinuity in the Δr function which represents a real two-point contact, such a contact geometry is considered to be less than optimum since two-point contact occurs in the small radius curves, which increases in particular the guiding forces and thus also the wheel and rail wear.

The Δr function is the starting point for the evaluation of the contact conditions in curves. Essential values of this function are marked by the following points in Figure 10:

Point O - kinematic rolling position

The point O on the Δr function represents the lateral offset in which the wheelset in the curve can roll purely kinematically in a radial position.

Point A^o

The point A^o denotes the bottom contact point of a pronounced discontinuity, which represents a two-point contact. The contact point corresponding to A^o is always on the tread of the wheel.

Point A^{oo}

The point A^{oo} denotes the upper contact point of a pronounced discontinuity, which represents a two-point contact. The contact point corresponding to A^{oo} is always on the wheel flange.

Point E

The point E lies on the Δr function at the lateral displacement $y_E = y_{A^o}$ - 1 mm. The definition of this point was made under the assumption that with 1 mm margin before the discontinuity point an ideal positioning of the wheelset (radial position) is possible. The margin permits the generation of steering forces to overcome the longitudinal forces in the suspension for a constrained wheelset.

Border between the wheel flange and the tread

In case there is no pronounced discontinuity before flange contact is reached then the point A^o is defined as the last point before |Δr| = 10 mm when starting at y = 0.

The described points are used to determine the following parameters:

Parameters d_E and e_E of the function Δr = f (y)

The parameter d_E is the distance between the points O and E with respect to the y-coordinate. The parameter e_E represents the distance of point E from point O with respect to Δr. Both parameters have already been described in [20]. In the present method the distance d is reduced by 1 mm (d becomes d_E). Thus, the ordinate value e_1 is read not at the point A^o but at the wheelset lateral displacement y = (y_{A^o}- 1 mm) at the point E (e_1 becomes e_E).

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Parameter q_{A^o}

The quotient q_{A^o} is the kinematically negotiable curve radius at the point A^o normalized with the nominal radius of the examined curve:

$$q_{A^o} = \frac{R_{A^o}}{R}$$

with

— $R_{A^o} = r_0 \frac{2b_A}{\Delta r_{A^o}}$ … radius of the kinematically negotiable curve corresponding to the rolling radius difference of point A^o;

— R … radius of the examined curve.

Radial steering index q_E

The radial steering index according to UIC 518 [22] and EN 14363 [23] is the kinematically negotiable curve radius at point E normalized with the radius of the examined curve:

$$q_E = \frac{R_E}{R}$$

with

a) R_E radius of the kinematically negotiable curve corresponding to the rolling radius difference of point E;

b) R radius of the examined curve.

That means that

c) if $q_E \leq 1$ radial steering is possible; and

d) if $q_E > 1$ radial steering is not possible but flange contact will occur before a rolling radius difference Δr, big enough for the curve in question, is achieved.

Rolling radii coefficient ρ_E

The rolling radii coefficient was introduced in the revised EN 15302 to allow for a clearer understanding of the practical meaning. It is defined as:

$$\rho_E = \frac{R}{R_E} - 1 = \frac{R}{r_0 \cdot 2b_A} \cdot \Delta r_E - 1$$

which means

e) if $\rho_E \geq 0$ radial steering is possible; and

f) if $\rho_E < 0$ radial steering is not possible, but flange contact will occur before a rolling radius difference Δr, big enough for the curve in question, is achieved.

The rolling radii coefficient can also be obtained from the radial steering index:

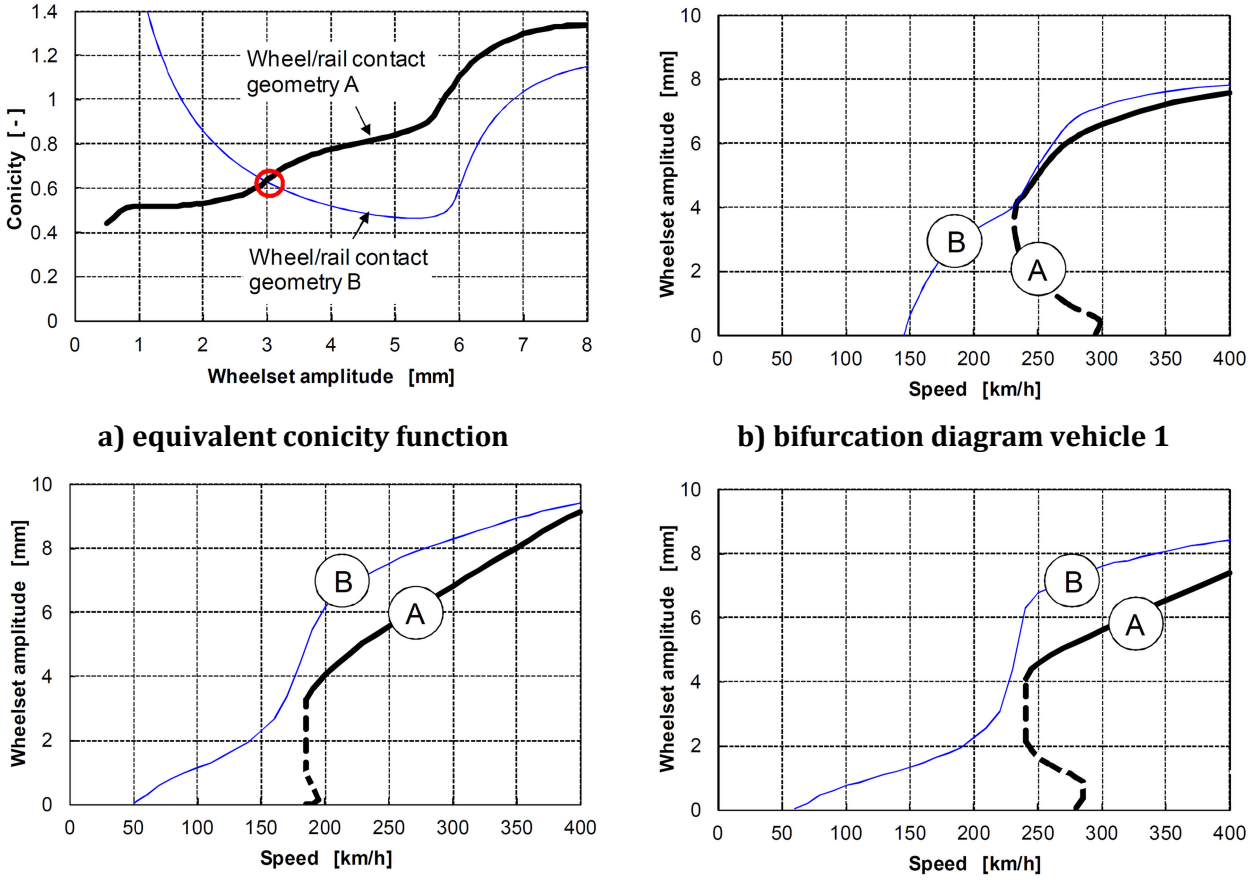
$$\rho_E = \frac{1}{q_E} - 1$$

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5.3 Nonlinearity parameter

Equivalent conicity is traditionally used to assess the contact geometry wheelset/track in regard to running stability. However, the equivalent conicity as a linearized parameter does not consider the nonlinearity of wheel-rail contact geometry.

An important difference of a nonlinear dynamic system compared to a linearized system is a possible coexistence of multiple solutions depending on system parameters. To visualize the multiplicity of solutions, bifurcation diagrams are used. For railway vehicles, a bifurcation diagram usually displays the amplitude of wheelset lateral oscillation versus speed. Investigations in [1] and [2] as well as comparisons with other publications resulted in the conclusion, that there is an interrelationship between the shape of the bifurcation diagram of a railway vehicle and the contact nonlinearity represented by the equivalent conicity as a function of wheelset displacement amplitude. This relationship can be seen in Figure 11 for three different vehicles and two different contact geometries wheelset-track. Both examples of wheel-rail contact geometry represent the same equivalent conicity for 3 mm amplitude, but different values for other wheelset amplitudes.



a) equivalent conicity function

b) bifurcation diagram vehicle 1

c) bifurcation diagram vehicle 2

d) bifurcation diagram vehicle 3

Figure 11 — Effect of the shape of the equivalent conicity function on the bifurcation diagrams of three different vehicles (from [2])

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A newly proposed nonlinearity parameter N_P [2] describes the presented property of the contact geometry wheelset-track. It is defined as the slope of the conicity function between the conicity value λ_2 for the wheelset displacement amplitude of 2 mm and conicity value λ_4 for the wheelset displacement amplitude of 4 mm:

$$N_P = \frac{\lambda_4 - \lambda_2}{2}$$

If a railway vehicle does not contain significant nonlinearities in its suspension such as friction components or a play in the wheelset guidance, the vehicle behaviour is predominantly influenced by the nonlinearities of the wheel-rail contact. This effect of contact geometry wheelset-track on the shape of the bifurcation diagram and the vehicle's behaviour at the stability limit is illustrated in Figure 12 using two wheel-rail contact geometries with the same equivalent conicity for a wheelset displacement amplitude of 3 mm but different values of the nonlinearity parameter: a positive N_P for contact geometry A (top diagrams) and a negative N_P for contact geometry B (bottom diagrams).

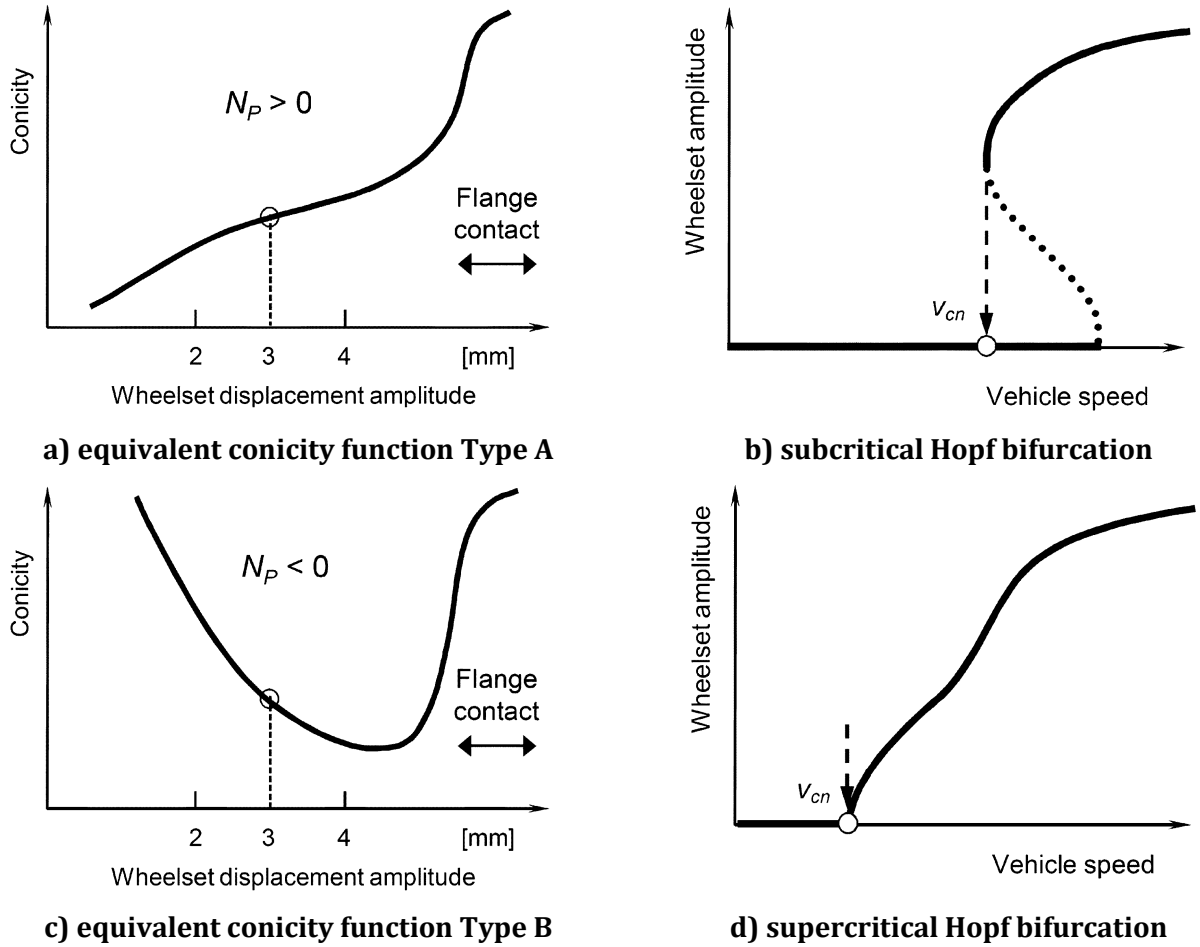
The effect of N_P on the vehicle behaviour can be described as follows:

— A positive N_P results in a sudden change between stable running and limit cycle oscillations. This expresses itself in so called subcritical Hopf bifurcation (top right diagram in Figure 12). Once the limit oscillation starts, it achieves large amplitudes. Moreover, two solutions co-exist in the speed range of the unstable saddle-cycle attractor displayed as a dotted line in the bifurcation diagram: a stable solution without oscillations and a limit cycle. Nonlinear critical speed v_{cn} is the lowest speed at which the self-excited oscillation can appear.

— A negative N_P results in a gradual change of vehicle behaviour at the stability limit. The limit cycle starts at a lower speed v_{cn} than in the previous case. The oscillation starts with a small amplitude which increases with increasing speed; the bifurcation diagram shows supercritical Hopf bifurcation (bottom right diagram in Figure 12).

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a) equivalent conicity function Type A

b) subcritical Hopf bifurcation

c) equivalent conicity function Type B

d) supercritical Hopf bifurcation

Figure 12 — Interrelationship between the type of conicity function (left diagrams) and the type of Hopf bifurcation (right diagrams)

The nonlinearity parameter N_P influences not only the vehicle behaviour at the stability limit but also the vehicle dynamics at lower speeds, below the critical speed. The contact geometries wheelset-track with a negative value of N_P lead under the same speed and the same track quality conditions to higher lateral forces between wheel and rail and larger lateral oscillations than the contact geometries wheelset-track with a positive N_P value [2] and [3].

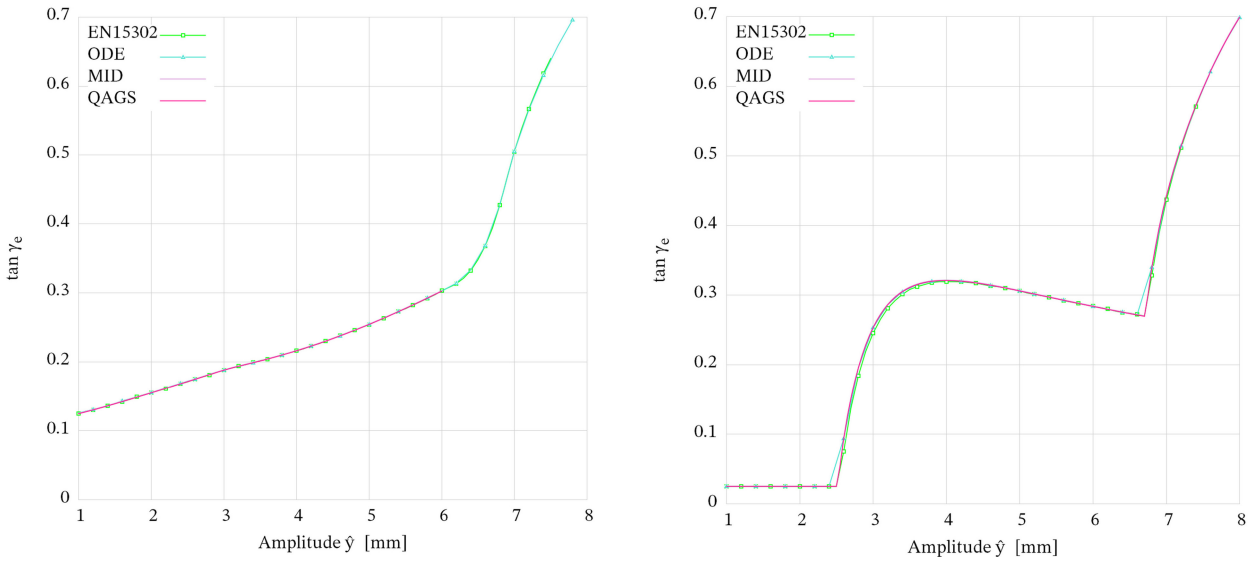
5.4 Calculation of equivalent conicity by two-step integration

EN 15302:2021, Annex D presents a method for calculation of the wavelength by two-step integration of the nonlinear differential equation of the lateral wheelset motion.

Another possible derivation leading to the same two integrals follows from the energy conservation in an undamped oscillator (see Kneubühl [38]). At the first step the system's potential energy is computed by integration, and in a second step an improper integral with singularities at both bounds has to be evaluated.

For the improper integral, in some cases a closed form solution is possible, but in general algorithms for numerical quadrature have to be employed. Due to potential problems at the integration boundaries, methods with adaptive step size control should be used. Using a typical constant stepsize of the Δr function will lead to incorrect results.

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a) equivalent conicity rail A/wheel A **b) equivalent conicity rail A/wheel H**

Figure 13 — Quadrature illustrated by means of the profile combinations of rail A with wheel A and rail A with wheel H

Figure 13 shows two examples comparing two quadrature methods (QUADPACK routines Midpoint (MID) and QAGS) with results obtained by the analytical solution (EN 15302) and the time integration described below (ODE). The Midpoint approach is easier to be implemented but slower than the QAGS routine.

5.5 Calculation of equivalent conicity by direct integration of the kinematic equation of motion

The wheelset motion is characterized by a nonlinear homogeneous second order ordinary differential equation (ODE). As initial condition, the maximum lateral displacement $y = y_{emax}$ at speed $\dot{y} = 0$ can be chosen. By extending the state variables by their first derivatives, the ODE can be transformed into a system of first order ODEs.

For the efficient numerical solution of this type of problem, there is a wide variety of numerical methods. Single-step methods like the class of Runge-Kutta methods often feature good error estimation and thus enable an adaptive stepsize control, where the estimated integration error is used to adjust the stepsize after each iteration.

5.6 Calculation of equivalent conicity by harmonic linearization

For the calculation of the dynamic behaviour of nonlinear systems, the method of harmonic linearization has proven to be a particularly important tool. This method is commonly used to determine the parameters representing the nonlinear contact geometry wheelset-track in linearized computation models. In contrast to other methods, this method is not based on the wavelength of the periodic movement of the wheelset, but on the overall behaviour of the wheelset in track assuming a harmonic wheelset movement.

The contact geometry wheelset-track influences the movement of the wheelset in track by the kinematic centring due to the difference of the rolling radii, by the gravitational centring due to the difference of the contact angles and by the roll rotation of the wheelset around the longitudinal axis.

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These properties of the contact geometry wheelset-track can be characterized by three geometric describing functions [6, 7]:

The function of the wheelset rolling radius difference:

$$\Delta r(y) = r_1(y) - r_2(y)$$

The function of the contact angle difference

$$\Delta \tan\gamma(y) = \tan\gamma_1(y) - \tan\gamma_2(y)$$

The function of the wheelset roll angle

$$\varphi(y)$$

The aim of harmonic linearization is to find dynamically equivalent linear approximations of the three nonlinear functions for given amplitudes of the wheelset lateral movement. As these linear approximations are related to the particular wheelset amplitude, this process is also called "quasi-linearization".

To linearize each of the describing functions [8] requires a linear approximation $q = ky$ such that the expectation of the quadratic error $E(\Delta^2)$ between the nonlinear function $f(y)$ and the sought linear approximation $q = ky$

$$E(\Delta^2) = (f(y) - ky)^2$$

takes a minimum. This means a search for extremum, which leads to the condition:

$$\frac{\partial}{\partial k} E(\Delta^2) = 0$$

After performing the differentiation, the linearization coefficient k is given by:

$$k = \frac{E(y \cdot f(y))}{E(y^2)}$$

The approach of harmonic wheelset movement with an amplitude $\hat{y}$ is then applied:

$$y(\tau) = y_0 + \hat{y}\sin\tau$$

And finally, by means of integration over a period, the "describing function" $k(\hat{y})$ is obtained which is dependent on the amplitude $\hat{y}$ of the harmonic linearization:

$$k(\hat{y}) = \frac{\int_0^{2\pi} (y_0 + \hat{y}\sin\tau)\ f(y_0 + \hat{y}\sin\tau)\, d\tau}{\int_0^{2\pi} (y_0 + \hat{y}\sin\tau)^2\, d\tau}$$

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The solution of the integral in the denominator and the substitution of the integration-differentials leads to the following representation of the describing function $k(\hat{y})$:

$$k(\hat{y}) = \frac{1}{\pi \hat{y}} \int_0^{2\pi} f(y_0 + \hat{y} \sin\tau) \sin\tau \, d\tau$$

Normalizing the contact angle difference and roll angle functions with the distance b_A (a half of the nominal contact point spacing) leads to the following formulations for the calculation of the three linearization parameters [7]:

The equivalent conicity

$$\tan\gamma_e = \frac{1}{2\pi \hat{y}} \int_0^{2\pi} \Delta r(y_0 + \hat{y} \sin\tau) \sin\tau \, d\tau$$

The normalized contact angle parameter

$$\varepsilon_e = \frac{b_A}{2\pi \hat{y}} \int_0^{2\pi} \Delta \tan\gamma(y_0 + \hat{y} \sin\tau) \sin\tau \, d\tau$$

The normalized roll angle parameter

$$\sigma_e = \frac{b_A}{\pi \hat{y}} \int_0^{2\pi} \varphi(y_0 + \hat{y}\sin\tau) \sin\tau \, \mathrm{d}\tau$$

The harmonic linearization is the only possibility to determine all three linearization parameters required for linearized calculations. This linearization method is usually available in simulation tools and is thus used also for the determination of equivalent conicity of measured profiles of wheels and rails. The equivalent conicity values for a wheelset amplitude of 3 mm evaluated by harmonic linearization are often the same or similar to the values calculated by the integration methods using the wavelength of the kinematic wheelset motion. A difference can occur particularly if there is a discontinuity of the rolling radius difference function close to the evaluated amplitude, see examples of comparisons in [2].

5.7 Updated reference profiles and results based on analytical solutions

All reference profiles of the wheels A, B, C, H and I and the rail profile A given in EN 15302 are defined by concatenations of linear sections and circular arcs. Annex I of EN 15302 contains the analytical definitions as well as tabular data of all reference profiles. The transitions between those sections are always continuous but may not be differentiable, e.g. at the transition between the two different slopes of profile H. This required the specification of all parameters, for instance the coordinates of the arc centres, with sufficient precision (at least 5 decimal places in the tabular data) and therefore the definitions have been updated. Numerical studies have shown that a change of the arc centre by 1e-5 mm can yield a significant contact-point jump.

In addition to the wheel profiles defined so far, the new wheel profile C was added to the list of reference profiles which is based on the widely used profile EPS (see EN 13715). It contains a cylindrical section in the running surface, similar to profile I.

The analytical solutions for the calculation of the wavelength and thus the equivalent conicity all assume the existence of an antiderivative in the double integration over the oscillation period. This is the case only for a certain class of Δr functions. Apart from the trivial linear case – whose solution is the “Klingel” formula – a cubic function yields the undamped Duffing equation (see Kneubühl [38]) with the Jacobi

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